Appendix D: Dry Cooling

Introduction

This appendix presents a synopsis of the design and operation of an air-cooled condenser system (dry cooling) and its applicability for existing power plants. The majority of the background information included in this discussion on dry cooling is from references listed at the end of this appendix, the background discussion draws primarily from Burns and Michiletti, 2000.

Dry cooling systems transfer heat to the atmosphere without the evaporative loss of water. There are two types of dry cooling systems for power plant applications: direct-dry cooling and indirect-dry cooling. Direct-dry cooling systems utilize air to directly condense steam, while indirect dry cooling systems utilize a closed-cycle water cooling system to condense steam, and the heated water is then air cooled. In the Agency's determination, indirect-dry cooling generally is the only application of the technology that would be considered for retrofit situations at existing power plants because a condenser would already be in place for a once-through or recirculated cooling system. For dry cooling towers the turbine exhaust steam exits directly to an air-cooled, finned-tube condenser. In the Agency's view, if this application would be applied to an existing plant, the entire steam turbine would necessarily be replaced or reconfigured in an unprecedented fashion. The costs of steam turbines are significantly more expensive than any type of recirculating cooling system, including the dry cooling systems. The Agency has determined that the feasibility of direct-dry cooling systems for existing plants is not demonstrated and because of the limitations and potential costs is not a candidate for retrofit situations. Therefore, the Agency does not further consider direct-dry cooling systems for existing facilities, though they are referred to significantly throughout the remainder of this appendix. Because direct-dry cooling systems would be more efficient and less costly than indirect-systems (ignoring the feasibility issues addressed above), the Agency's analyses of dry cooling systems using direct-dry cooling systems would show increased energy penalties and signficantly higher costs.

For indirect-dry cooled systems, recirculating fluid (usually water) passes through an air-cooled, finned tube tower. In contrast to direct-dry cooling, indirect-dry cooling does not operate as an air-cooled condenser. In other words, the steam is not condensed within the structure of the dry cooling tower, but instead indirectly through an indirect heat exchanger (that is, a surface condenser). Therefore, the indirect-dry cooling system would need to overcome additional heat resistance in the shell of the condenser compared to the direct dry cooling system. Ultimately, the inefficiency penalties of indirect dry cooling systems will exceed those of direct-dry cooling systems in all cases. Similar to the direct-dry cooling systems, the arrangement of the finned tubes are most generally of an A-frame pattern (which reduces the land area required compared to other configurations). However, due to the fact that dry cooling towers do not evaporate water for heat transfer, the towers are quite large in comparison to similarly sized wet cooling towers. Additionally, because indirect-dry cooled systems also utilize a surface condenser, with additional heat transfer inefficiencies compared to a direct-dry cooled system, the indirect systems are generally considered to be significantly larger than direct systems for comparable heat loads. Because dry cooling towers rely on sensible heat transfer, a large quantity of air must be forced across the finned tubes by fans to improve heat rejection. The number of fans is therefore considerably larger than would be used in a mechanical-draft wet cooling tower.

The key feature of dry cooling systems is that no evaporative cooling or release of heat to surface water occurs. As a result, water consumption rates are very low compared to wet cooling systems. Since the unit does not rely in principle on evaporative cooling as does a wet cooling tower, larger volumes of air must be passed through the system compared to the volume of air used in wet cooling towers. As a result, dry cooling towers need larger heat transfer surfaces and, therefore, tend to be larger in size than comparable wet cooling towers. The design and performance of the dry cooling system is based on the ambient dry bulb temperature. The dry bulb temperature is higher than the wet bulb temperature under most circumstances, being equal to the wet bulb temperature only when the relative humidity is at 100%.

Direct-dry cooling has been installed at a variety of power plants utilizing many fuel types. In the United States, dry cooling is most frequently applied at plants in northern climates. Additionally, arid areas with significant water scarcity concerns have also experiencing growth in dry cooling system projects. However, each of the demonstrations that the Agency has studied is for a direct-dry cooling system configured for a new facility project. As demonstrated in Chapter 5, the comparative energy penalty of a direct-dry cooling plant in a hot environment at peak summer conditions can exceed 12 percent. Additionally, the indirect-dry cooling system would be even less efficient, producing maximum energy penalties of 18 percent according to the Department of Energy (DOE, 2001). Additionally, indirect-dry cooling systems would likely cause prohibitively high exhaust turbine backpressures, thereby potentially debilitating the operation of some plants at peak-summer, peak-demand conditions (DOE, 2001).

As with wet cooling towers, the ambient air temperature and system design can have an effect on the steam turbine exhaust pressure, which in turn affects the turbine efficiency. Thus, the turbine efficiency can change over time as the air temperature changes. The fans used to mechanically force air through the condenser represent the greatest operational energy requirement for dry cooling systems.

A design measure comparable to the approach value used in wet towers is the difference between the design dry bulb temperature and the temperature of saturated steam at the design turbine exhaust pressure. In general, for direct-dry cooling systems a larger, more costly dry cooling system will produce a smaller temperature difference across the dry cooling tower and, therefore, a lower turbine exhaust pressure. However, as calculated by DOE, the 40 degree F approach indirect-dry cooling towers may actually be less efficient than smaller sized 20 degree F towers in the cases modeled by DOE.

Steam turbines are designed to operate within certain exhaust pressure ranges. In general, steam turbines that are designed to operate at the exhaust steam pressure ranges typical of wet cooling systems, which generally operate at lower exhaust pressures (e.g., <5 in Hg), may be damaged if the exhaust pressure exceeds a certain value. Even the highest values of operable exhaust pressures may be exceed with retrofitted indirect-dry cooling systems. For existing facilities, many with aged turbines, this is a fundamental engineering problem for the feasibility of the retrofitted dry cooling system.

In an analysis for the New Facility rule, EPA examined turbine exhaust pressures at the highest design dry bulb temperatures in the U.S. (which were around 100 °F), which ranged from 5.0 to 9.5 inches Hg. The highest value of 9.5 inches Hg was for a refinery power system in California which, based on the steam rate, was comparable to other relatively small systems generating several megawatts and apparently did not warrant the use of an efficient cooling system. The other data showed turbine exhaust pressures of around 6 to 7 inches Hg at dry bulb temperatures of around 100 °F. Maximum exhaust pressures in the range of 8 to12 inches Hg may be expected in hotter regions of the U.S.(Hensley 1985). An air cooled condenser analysis (Weeks 2000) reports that for a combined cycle plant built in Boulder City, Nevada, the maximum ambient temperature used for the maximum off-design specification was 108 °F with a corresponding turbine exhaust pressure of 7.8 inches Hg. Note that the equation used by EPA to generate the turbine exhaust pressure values in the energy penalty analysis produced an estimated exhaust pressure of 8.02 inches Hg at a dry bulb temperature of 108 °F. For wet towers, the typical turbine exhaust pressure operating range is1.5 to

3.5 inches Hg(Woodruff 1998). EPA prepared all of this analysis in the context of direct-dry cooling systems installed at new facility projects. For hypothetical indirect-dry systems at existing facilities, the backpressures would be significantly higher than the values examined by the Agency for the New Facility rule.

In addition, the issue of demonstrated dry cooling systems (even for new facilities) emerges in the context of fossil-fuel and nuclear plants. The largest operating coal-fired plant in the United States with dry cooling is the Wyodak Station in Gillette, WY with a total cooling capacity of 330 MW (1.88 million lb/hr of steam). EPA notes that this is significantly smaller than a vast number of coal-fired plants within the scope of this proposal. In addition, the design temperature of the direct-dry cooled system at this plant (which directly affects the size of the dry cooling system) is below average for summer conditions throughout the United States (the Wyodak Station has a design temperature of 66 deg F). The Agency reiterates its reservation of applying requirements based on dry cooling at the sizes of coal-fired and nuclear plants in the scope of this proposal.

Costs of Dry Cooling

For the New Facility Final Rule, the Agency projected that the total annualized cost for the dry cooling alternative was $490 million (in 2000 dollars) for 121 facilities. This proposed rule applies to 539 facilities, and therefore, a regulatory option based on dry cooling for all plants would impose a dramatically higher annual compliance cost than that estimated for the New Facility Final Rule. In addition, the costs the dry cooling system would be even more dramatic, due to the fact that the majority of existing facilities within the scope of this rule operate with once-through cooling systems, whereas for the New Facility Final Rule, the vast majority of plants were projected to install recirculating wet cooling at baseline, thereby reducing marginal cost increases.

Although the dry cooling option is extremely effective at reducing impingement and entrainment and would yield annual benefits of $138.2 million for impingement reductions and $1.33 billion for entrainment reductions at existing facilities, it does so at a cost that would be unacceptable. EPA recognizes that dry cooling technology uses extremely low-level or no cooling water intake, thereby reducing impingement and entrainment of organisms to dramatically low levels. However, EPA interprets the use of the word “minimize” in section 316(b) in a manner that allows EPA the discretion to consider technologies that very effectively reduce, but do not completely eliminate, impingement and entrainment and therefore meet the requirements of section 316(b). Although EPA has rejected dry cooling technology as a national minimum requirement, EPA does not intend to restrict the use of dry cooling or to dispute that dry cooling may be the appropriate cooling technology for some facilities. For example, facilities that are repowering and replacing the entire infrastructure of the facility may find that dry cooling is an acceptable technology in some cases. A State may choose to use its own authorities to require dry cooling in areas where the State finds its fishery resources need additional protection above the levels provided by these technology-based minimum standards.

Methodology for Dry Cooling Cost Estimates at Existing Facilities

For the purposes of approximating the hypothetical costs of retrofitting to dry cooling for existing facilities, the Agency used the following methodology for developing facility-level costs estimates:

- Capital costs for the dry cooling towers were estimated using the cost equation that was developed for the New Facility Rule (see the next section for the New Facility dry cooling cost estimates).
- The cost equations are based on equivalent cooling water flow rates (gpm) using the once-through design intake cooling flow as the independent variable. To aviod using the equation outside of its valid range, for facilities with

intake flows greater than 225,000 gpm (which is the maximum equation input value plus 10%), costs for multiple equal size "units" were developed and then added together.

- An additional 5 percent was added to the capital costs as an "allowance" for unforseen costs.
- A cost factor of 5 percent was added to the dry tower capital costs to account for retrofit costs.
- Intake pumping was assumed to decrease to zero or near zero. Therefore, no costs are included for intake or piping modifications.

The dry cooling capital cost equation is shown below:

Capital Cost (Dollars) = $-0.00000000008*(\text{gpm})^3 + 0.0001*(\text{gpm})^2 + 189.77*(\text{gpm}) + 800490$

Note that the capital costs do not include any consideration for replacement or modification of the steam turbines. Nor do the O&M costs below include consideration of the effects on turbine efficiency resulting from the differences in turbine exhaust pressure caused by changes in the cooling system.

<u>Dry Cooling O&M Costs</u>

EPA has revised the O&M costs using a different basis than was used for the New Facility Rule compliance cost estimates. Rather than base the costs on factors applied to the capital cost as was previously done, EPA based the O&M cost on energy requirements and cost information obtained from facility personnel and an air-cooled condenser manufacturer.

O&M cost components include the following:

- Labor costs starting at $12,000/yr for a 2,000 gpm equivalent system increasing to a maximum of two full time maintenance personnel (at a salary of $55,250/yr) for a 204,000 gpm equivalent system.
- Fan energy costs are based on 800 gpm/MW and $6,000/MW. The $6,000/MW value is based on EPA's estimated fan energy penalty of 2.4% plus an annual operating duration of 7860 hours and an average electricity value of $30/MW-hr**.
- Costs for grease, oil, and high pressure spray cleaning starting at $500/yr for a 2,000 gpm equivalent system increasing to $19,500 for a 204,000 gpm equivalent system.
- Costs for blade replacement, gaskets and other minor items not covered by warranty starting at $3,000/yr for a 2,000 gpm equivalent system increasing to $9,600/yr for a 204,000 gpm equivalent system.
- Since intake volumes are reduced to near zero, post-compliance monitoring costs are assumed to be zero.

The dry cooling O&M cost equation is shown below:

O&M (Dollars) = $53.122*(\text{gpm})^{0.8442}$

A dry cooling system manufacturer has indicated that major components including the air cooled condenser and fan motors should not require replacement over the 30 yr life of the equipment.

** the average electricity price of $30 per MWh is a combination of the energy price and the capacity price for the 530 in-scope facilities modeled by the Integrated Planning Model (IPM). It is a weighted average of the 530 facilities, based on the 2008 IPM base case run using the EPA electricity demand growth projections.

Methodology for Dry Cooling Cost Estimates at New Facilities

EPA estimated the capital costs using relative cost factors for various types of wet towers and air cooled condensers (that is, direct-dry cooling systems), using the cost of a comparable wet tower constructed of Douglas Fir as the basis. EPA used cost factors developed by industry experts who manufacture, sell and install cooling towers, including air cooled systems, for power plants and other applications. EPA based the capital costs on these factors with some modifications. To be conservative, EPA chose the highest value within each range as the basis. The factors chosen are 325 percent and 225 percent (of the cost of a mechanical wet tower) for capital cost (for a tower with a delta of 10 ºF) and O&M cost, respectively. EPA applied a multiplier of roughly 1.7 to the dry tower capital cost estimates for a delta of 10 ºF to yield capital cost estimates for a dry tower with a delta of 5 ºF. EPA applied these factors to the capital costs derived for the basic steel mechanical draft wet cooling towers to yield the capital cost estimates for dry towers.

Note that the source document for the factors forming the basis of the estimates states that the factors represent comparable cooling systems for plants with the same generated electric power and the same turbine exhaust pressure. Since the cost factors generate equivalent dry cooling systems, the tower costs can still be referenced to the corresponding equivalent cooling water flow rate of the mechanical wet tower used as the cost basis. Since the §316(b) analyses focuses primarily on water use, the use of the cooling flow or the "equivalent" was considered as the best way to compare costs. The costing methodology uses an equivalent cooling water flow rate as the independent input variable for costing dry towers.

Using the estimated costs, EPA developed cost equations using a polynomial curve fitting function. Table 1 presents capital cost equations for dry towers with deltas of 5 and 10 degrees.

Table 1. Capital Cost Equations of Dry Cooling Towers with Delta of 5 °F and 10 °F

Delta	Capital Cost Equation[1]	Correlation Coefficient
5 ºF	$y = -2E\text{-}10x^3 + 0.0002x^2 + 337.56x + 973608$	$R^2 = 0.9989$
10 ºF	$y = -8E\text{-}11x^3 + 0.0001x^2 + 189.77x + 800490$	$R^2 = 0.9979$

1) x is for flow in gpm and y is cost in dollars.

Validation of Dry Cooling Capital Cost Curves

To validate the dry tower capital cost curves and equations, EPA compared the costs predicted by the equation for dry towers with delta of 10 ºF to actual costs for five dry tower construction projects provided by industry representatives. To make this comparison, EPA first needed to estimate equivalent flows for the dry tower construction project costs. Obviously, as noted above, dry towers do not use cooling water. However, for every power plant of a given capacity there will, dependent on the selected design parameters, be a corresponding equivalent recirculating cooling water flow that would apply if wet cooling towers were installed to condense the same steam load.

EPA used the steam load rate and cooling system efficiency to determine the equivalent flow. Note that the heat rejection rate will be proportional to the plant capacity. EPA estimated the flow required for a wet cooling tower that is functionally equivalent to the dry tower by converting each plant's steam tons/hour into cooling flow in gpm using the following equations:

Steam tons/hr x 2000 lbs/ton x 1000 BTUs/lb steam = BTUs/hr
One ton/hr = 12,000 BTU/hr
BTUs/hr / 12000 = Tons of ice
Tons of Ice x 3 = Flow (gpm) for wet systems

Chart 4-2 presents a comparison of the EPA capital cost estimates for dry towers with delta of 10 ºF (with 25% error bars) to actual dry tower installations. This chart shows that EPA's cost curves produce conservative cost estimates, since the EPA estimates are greater than all of the dry tower project costs based on the calculated equivalent cooling flow rate for the actual projects.

Chart 4-1. Capital Costs of Dry Cooling Towers Versus Flows Of Replaced Wet Cooling Towers (5 &10 Degrees Delta)

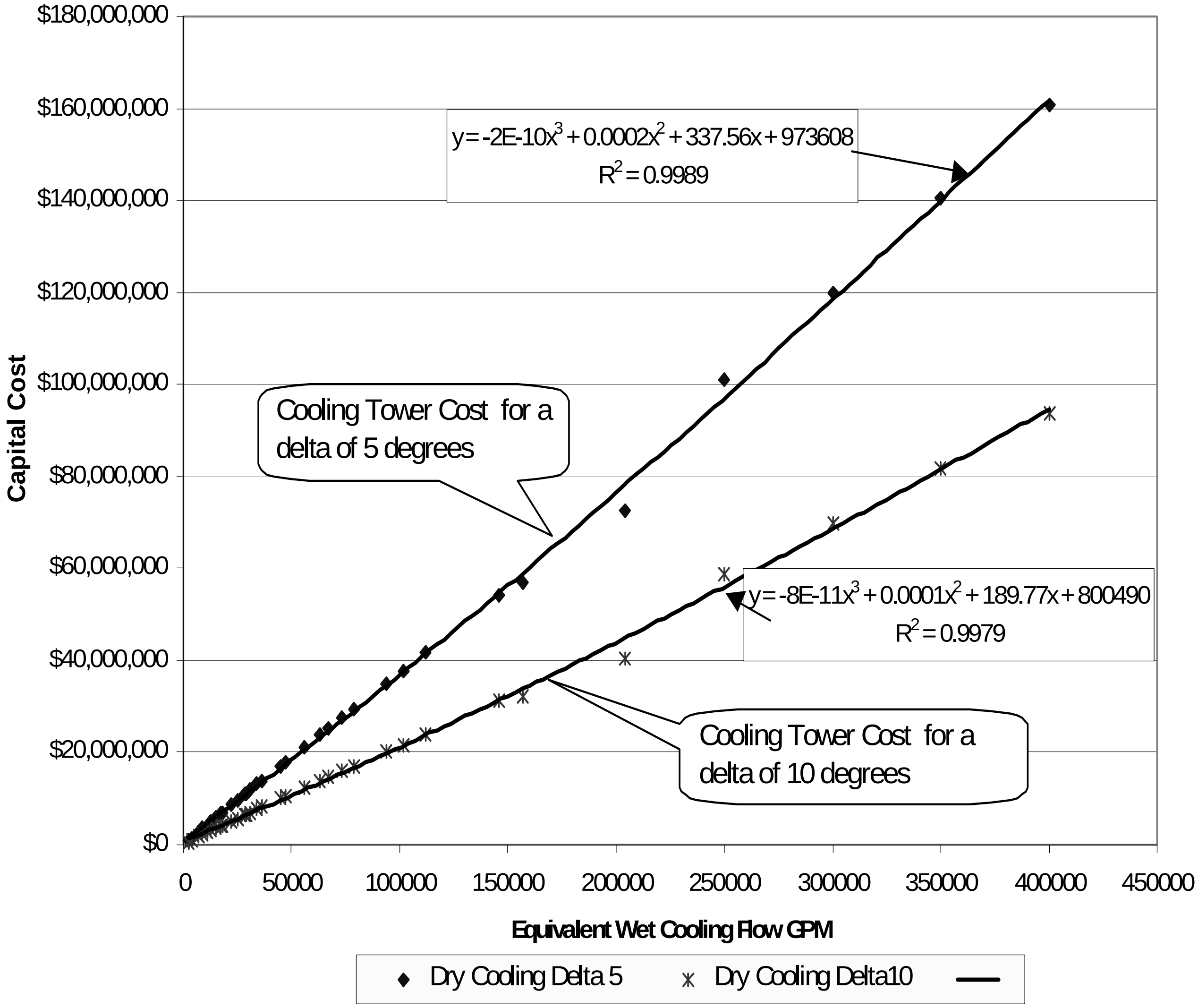
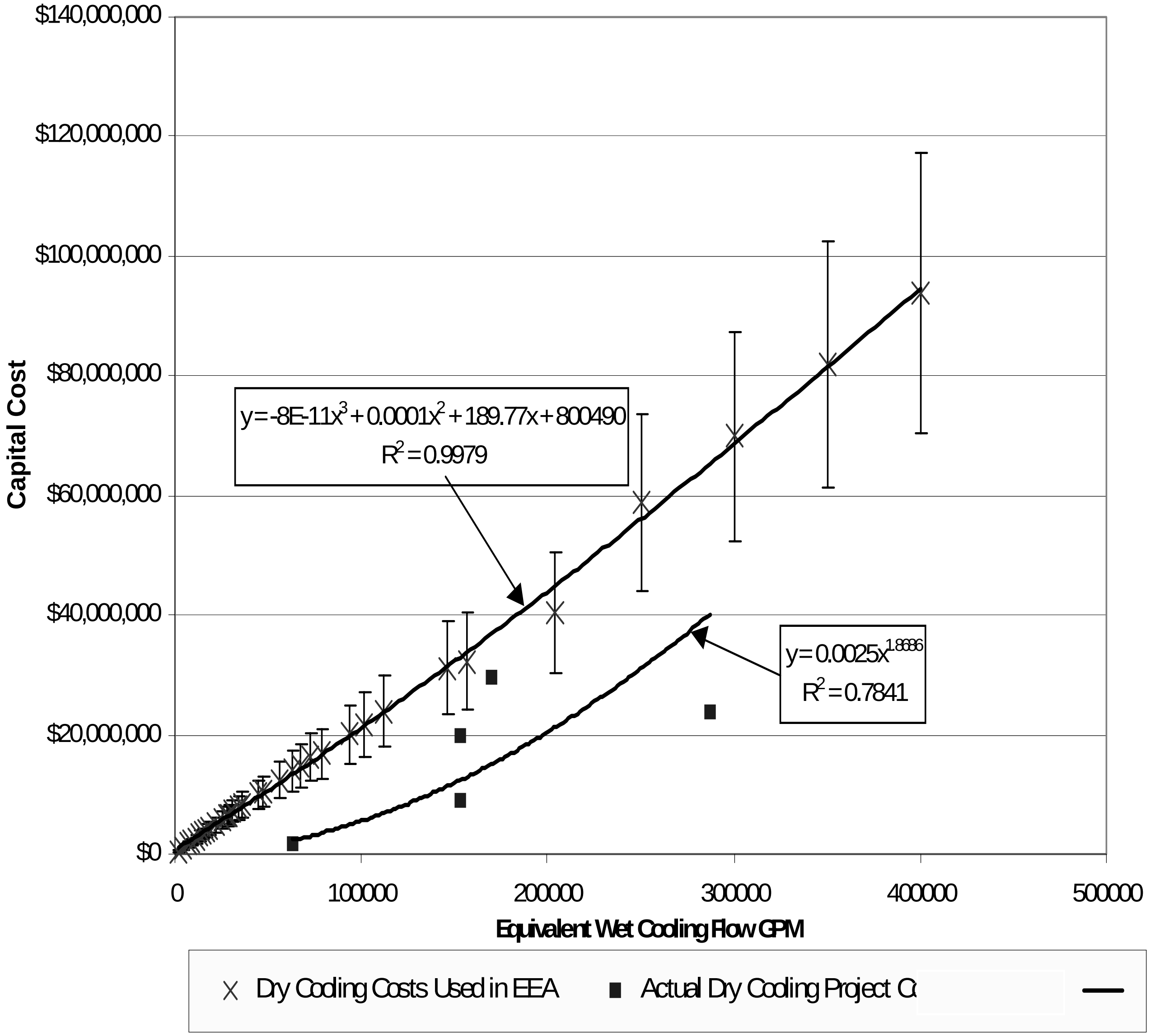


Chart 4-2. Actual Capital Costs of Dry Cooling Tower Projects and Comparable Costs from EPA Cost Curves



Evaluation of Dry Cooling as BTA

This section presents a summary of EPA's evaluation of the dry cooling technology as a candidate for best technology available to minimize adverse environmental impacts. Based on the information presented in the previous sections, EPA concluded that dry cooling systems do not represent the best technology available.

First, EPA concluded that dry cooling is not demonstrated nor likely feasible for the existing facilities within the scope of this proposed rule. As noted previously, indirect-dry cooling generally is the only application of the technology that would be considered for retrofit situations at existing power plants because a condenser would already be in place for a once-through or recirculating wet cooling system. As estimated by the DOE (2001), the comparative energy penalty of a retrofitted indirect-dry cooling plant in a hot environment at peak-summer conditions can approach 18 percent at a facility, thereby making dry cooling extremely unfavorable in many areas of the U.S. for some types of power plant types. Additionally, the predicted turbine backpressures of these systems may debilitate the operation of some plants, thereby severely disrupting energy supply and distribution.

In addition, EPA evaluated a regulatory option for dry cooling systems, based on favorable cost assumptions and concluded that the costs of dry cooling systems are prohibitively high in comparison to the benefits.

In summary, EPA concluded that dry cooling is not technically or economically feasible for the existing facilities potentially subject to this proposed rule, would increase air emissions due to dramatic energy penalties, and has a cost that is orders of magnitude more than requirements of this proposed rule. For these reasons, EPA concluded that dry cooling does not represent the "best technology available" for minimizing adverse environmental impact.

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